STRUCTURAL IMPROVEMENT PROPOSAL FOR THE TRUNNIONS MAINTENANCE OF ROTARY CAR DUMPERS *

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Abstract

Responsible for the removal of material from the train wagons and accommodation on conveyors belts, the Rotary Car Dumper is an essential equipment for handling bulk materials, and its stop by failure or even scheduled maintenance generates production losses. This paper proposes an improvement in the maintenance process of trunnions, responsible for dumper support, through structural modifications, in order to reduce the current 10 hours of production downtime to replacement the equalizing wheels. To this end, the proposed structure was modeled and characterized in SolidWorks software, and simulated using the Finite Element Method. Static and fatigue tests were performed according to the European Federation of Maintenance. Simulation results indicate that the change is feasible technically meeting the safety standards required by the FEM and its installation will not affect useful life of the system.

Keywords: Rotary car dumper; Finite element method; European standard FEM; Structural analysis.

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1 INTRODUCTION

In the process of mining during the transport and transfer by rail, systems of rotary car dumpers are used. This equipment comprises part of the discharge process of materials such as refined iron ore or iron ore pellets from the extraction mines and processing plants (Figure 1).

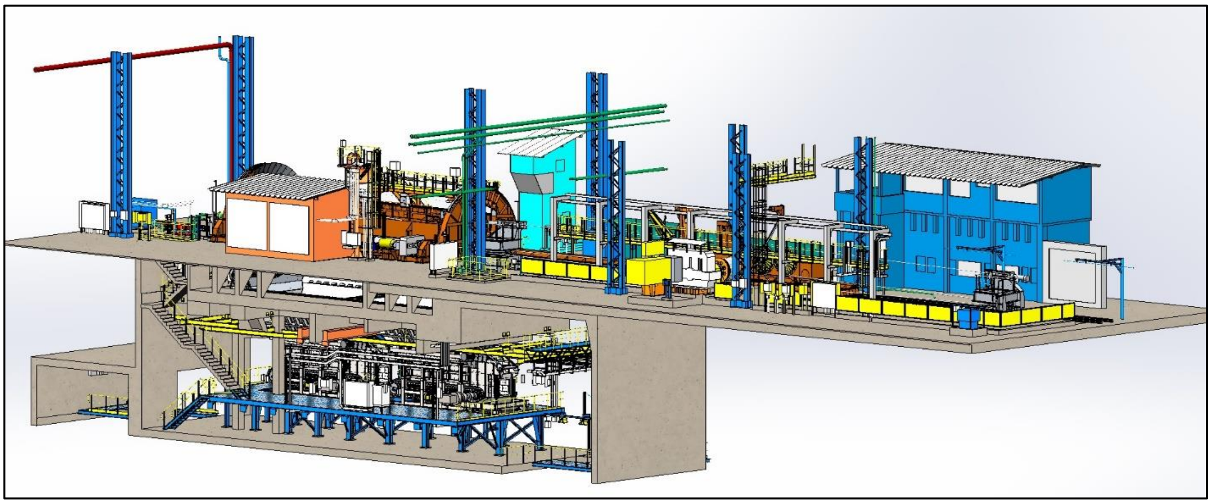

Figure 1. Location of rotary car dumpers.

The Car Dumpers have the function of receiving the wagons loaded with the material to be processed and discharges them through a swivel that can reach from 160º to 180º inside the hopper, accommodating the ore on the conveyor belts. Each rotary car dumper has a nominal design rate of 7,000 ton/h during its operating, the time of a discharge cycle is 79 s, and are composed of four different systems: dumper, hopper, clamps, pusher (Figure 2).

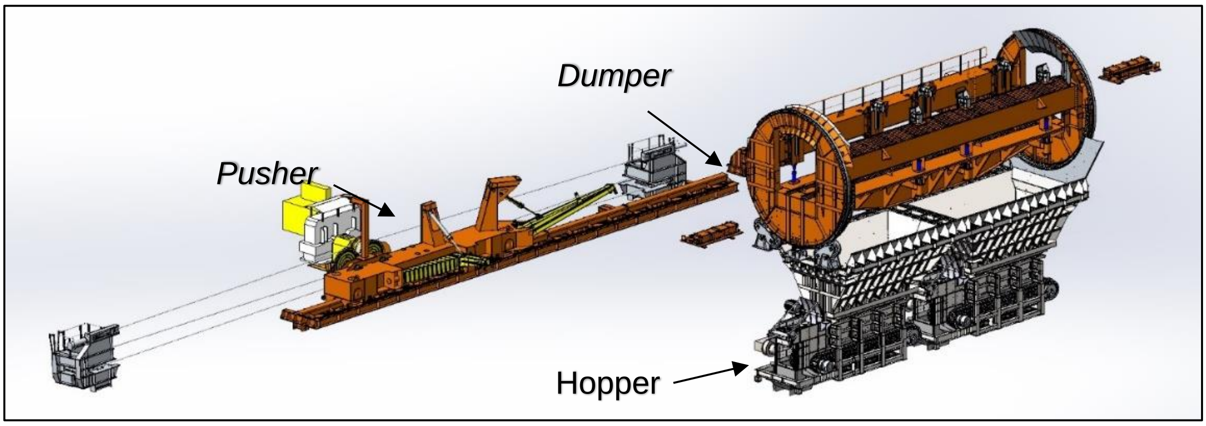

Figure 2. General arrangement.

The discharge of each pair of wagons is carried out by the dumper, being responsible for locking the pair of wagons in its interior, by the sides of the wagon through hydraulic clamps. The operational turning is timed in such a way that it allows all material to be dumped by gravity into the hoppers. As soon as the time expires, a reverse turn is performed so that the empty wagons return to the normal position at 0º, where the mechanical arm effects the positioning of the next pair of wagons inside the dumper, to start a new cycle.

The trunninons is a set belonging to the dumper system (Figures 3 and 4). The rotation is caused by operating the electric drive, which by means of a pinion transfers force to a rack screwed on the edges turning ring, thereby allowing rotational movement happen.

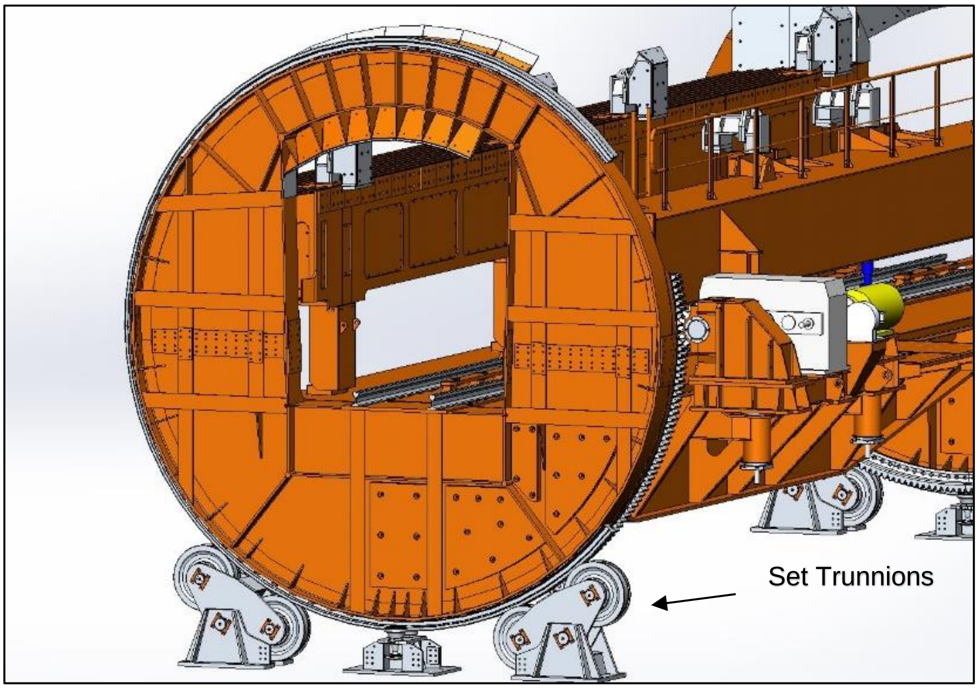

Figure 3. Turning ring of rotary car dumper.

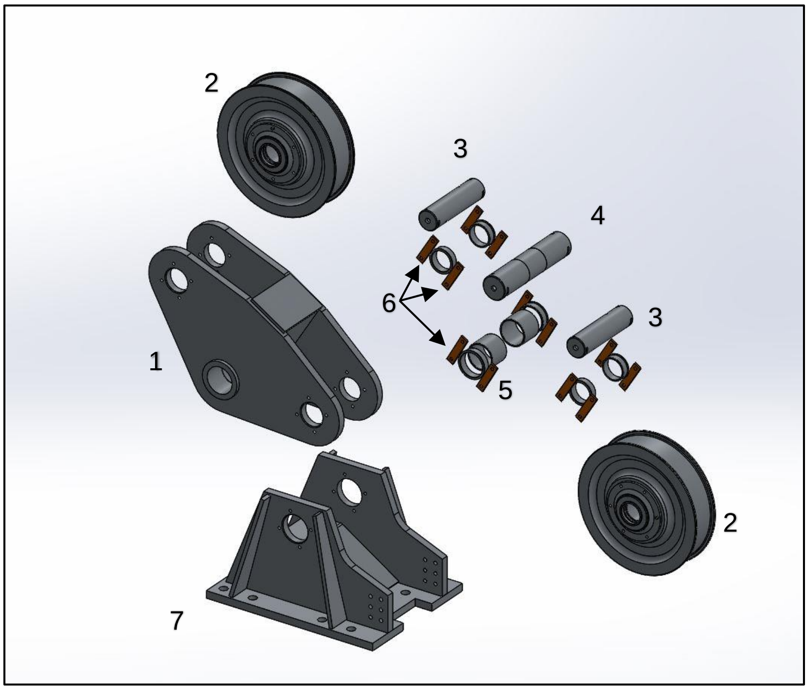

Figure 4. Vista do balancim explodido, componentes do conjunto do balancim.

The set trunnions (Figure 4) is constituted by: a balance (1); two equalizing wheels with machined friezes on the sides (2); two shafts (3) responsible for fixing the equalizing wheels on the balance; an shaft (4) responsible for positioning the balance with the equalizing wheels already mounted at an angle allowing contact with a rail installed in the turning ring; bushings ensuring rotation in the holes without wear to the shaft and structures (5); cotter pins so that there are no axial displacements in the shafts (6) and a structure which secures the whole assembly to the floor (7).

After 2 years, the life of the trunnions equalizer wheels comes to an end, requiring a replacement due to wear during its operation. Difficulties in maintaining the system are initiated during the movement of heavy parts in a reduced physical space (Figure 5), requiring extremely precise crane operation for the primary positioning of the equalizer wheel.

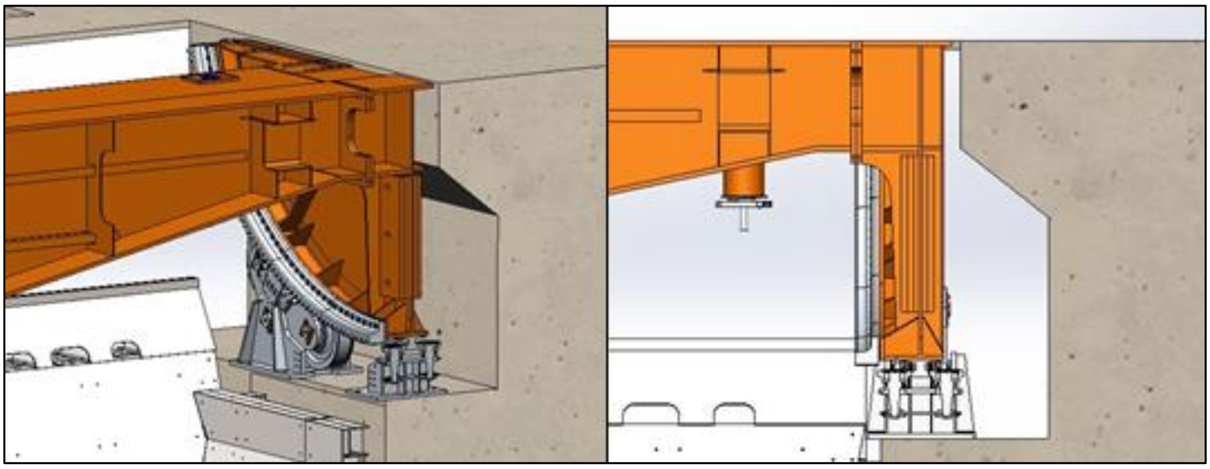

Figure 5. Cut view of trunnions installation site.

For an exchange of an equalizer wheel are required approximately 10 hours of intervention by the maintenance staff, the reduction in the time of operational availability is one of the complicating points to perform the work. Another point that has a direct impact on the equipment availability is the construction model of the trunnions, where the wheels are installed by shafts passing through holes in the balance structure.

During equalizer wheel exchange, the stage that demand a greater amount of time and that more deviations occur, is the shaft removal that pass through the equalizer wheel and balance. This is because the shaft has a 160mm diameter tolerance g8 (-14µm / -39µm) and the hole of the bushing installed in the structure of the balance has a diameter of 180mm with tolerance F8 (-43µm / -106µm), therefore, the gap present in the balance hole is less than the maximum tolerance in the shaft diameter.

One of the aggravating factors to reduce the useful life of the system is the inefficiency of the shaft lubrication system, caused by contamination of material in the lubricating channels and failure to comply with the lubrication plans prescribed by the equipment manufacturer.

The proposal of this article is to reduce the time of the operational stops for maintenance of the system through structural alterations in the constructive design of the trunnions, in addition to improving working conditions in view of physical limitations, increasing the safety of the teams involved in this intervention process in the equipment. Another limiting factor is the deficiency of physical space for the handling of specific tools for the execution of the activity.

2 DEVELOPMENT

Due to the difficulties encountered in the interventions performed in the Rotary Car Dumper, it was considered the alteration of the constructive design of the trunnions, adopting as a basic premise the removal of the axis of the equalizing wheel in a simple and practical way.

The change in design provides for the fixing of the equalizer wheel axle through bolted bearings in the balance structure (Figure 6), representing an ease in maintenance conditions while changing the equalizing wheel, since complications in the current project are characterized by the removal of the shaft in the axial direction.

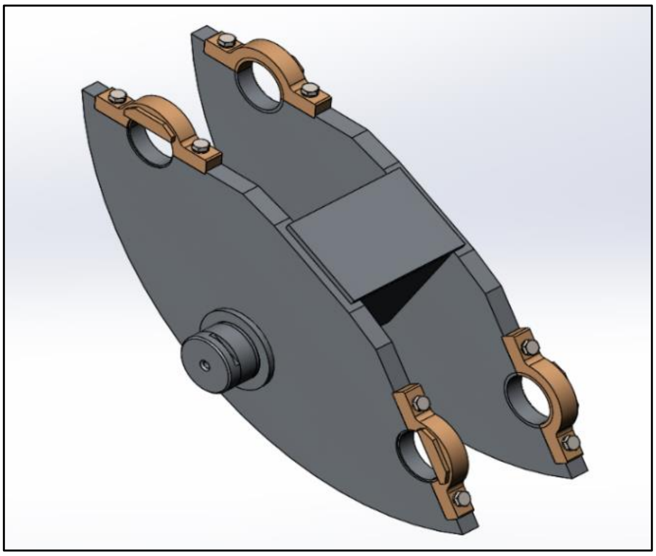

Figure 6. Trunnions after proposed changes.

2.1 Finite Element Method (FEM)

The FEM is a computational solution used for several types of engineering analyzes, since it is defined as a numerical methodology that occurs in continuous areas and that can describe some physical phenomena through numerical equations, which present high complexity of studied in an analytical way. [1].

Usually the human mind subdivides the systems into individual components or their elements that are part of the system [1]. Thus, the idea arises that, from the understanding of individual behavior, it is possible to understand the behavior of the whole set, however complex it may seem. The FEM divides the surfaces of a body into small subdomains, that is, it is discretized the whole area is interested in performing the finite element analysis. These elements are characterized by simple geometries such as squares, triangles or vertices for cases where the analysis will be two-dimensional, and prisms or quadrilaterals for cases where the analysis will be three-dimensional.

The development of the design and calculations were carried out in accordance with Norm FEM 2131/2132 Section II [2], which deals with the dimensioning of structures subject to traction, compression and fatigue in industrial equipment with continuous handling of bulk materials. The material used to manufacture the main structures of the rocker was the USI SAR 50, and has yield value $\sigma_{esc} = 345MPa\ for\ t \leq 57{,}15mm$, where, ‘t’ is the thickness of the plate [3].

2.1.1 Determination of the angle of the acting forces

In determining the angle of the flat surface of the bearing, it was not possible to apply a 90° angle to the normal passing through the center of the wheel, due to the difficulties that would be generated to remove the equalizing wheels from the assembly. Therefore, due to the difficulties observed for the maintenance of the rocker arms, it was decided to adopt the angle $\alpha_1 = 32°$ between the normal center of the wheel and the flat surface of the bearing face and $\alpha_2 = 7{,}8°$.

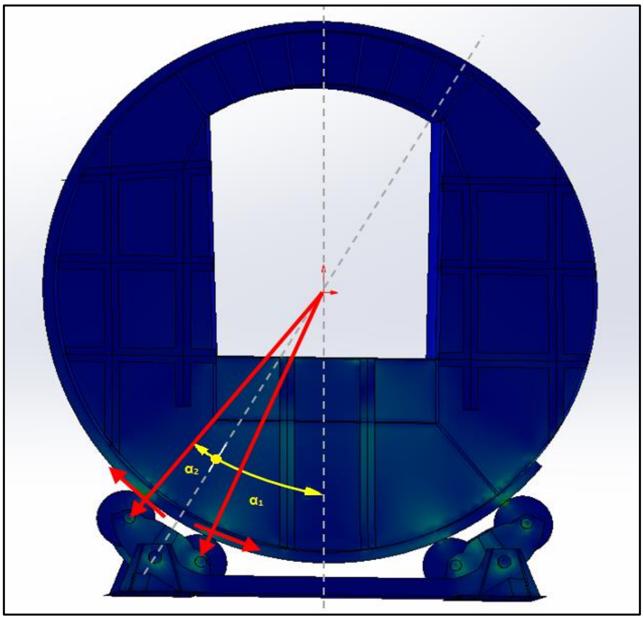

Figure 9. Angles of the forces acting on the trunnion used in the simulation.

2.1.2 Determination of loads acting

For calculation purposes, the adopted forces are the weight of the dumper structure, divided equally by 08 wheels equalizing. Accordingly, the weight acting on each wheel itself plus the mass was two wagons loaded kind of material circulating from railway. The acting loads used in the study are listed in Table 1.

Table 1. Loads acting on trunnions.

Loads	Inputs	Load acting			
		Total		Per Wheel	
		Kgf	kN	kgf	kN
Radial	Dumper Mass	182784,2	179,3	22848,0	224,1
	Mass of the Loaded Wagon	200000,0	196,2	25000,0	245,3
Tangential	Free Wheels	3403,0	33,4	425,4	4,2
	Locked Wheels	10209,0	100,2	1276,1	12,5
Axial	Considered in the original desing	18960,2	186,0	2370,0	23,3

2.2 Static Analysis of the Trunnions

The static analysis of tensions acting on the trunnion is necessary to verify the structural dimensioning, besides allowing to verify the tensions in the condition of the proposed changes. It was verified the static structural behavior of the trunnion next to the proposed bearing, and of the separate pieces. A total of 12 distinct conditions were simulated, the three most important are:

- Only dumper's own weight and unlocked equalizing wheels;
- The dumper's own weight loaded wagon wheels and equalizing unlocked;
- The dumper's own weight loaded wagon and locked equalizing wheels.

The analyzes were performed using a simplified model of the trunnion and bearing structure, but represent the constructive and operational reality of the system.

2.2.1 Trunnion and bearing with own weight of the dumper and free wheels

In this analysis, it was found that the higher stresses at Von Misses are concentrated in the region of bearing and balance attachment with the acceptable intensity of 164 MPa, which will be supported by the fastening element. In the rest of the structure, the tensions are uniform, as figure 10.

The FEM [2], according to Case I, indicates a minimum utilization rate (u) of 1.6:

$$u = \frac{345}{164} = 2{,}1 > 1{,}6$$

It can be concluded that only with the action of the own weight, the structures studied are in accordance with the standards of the FEM [2].

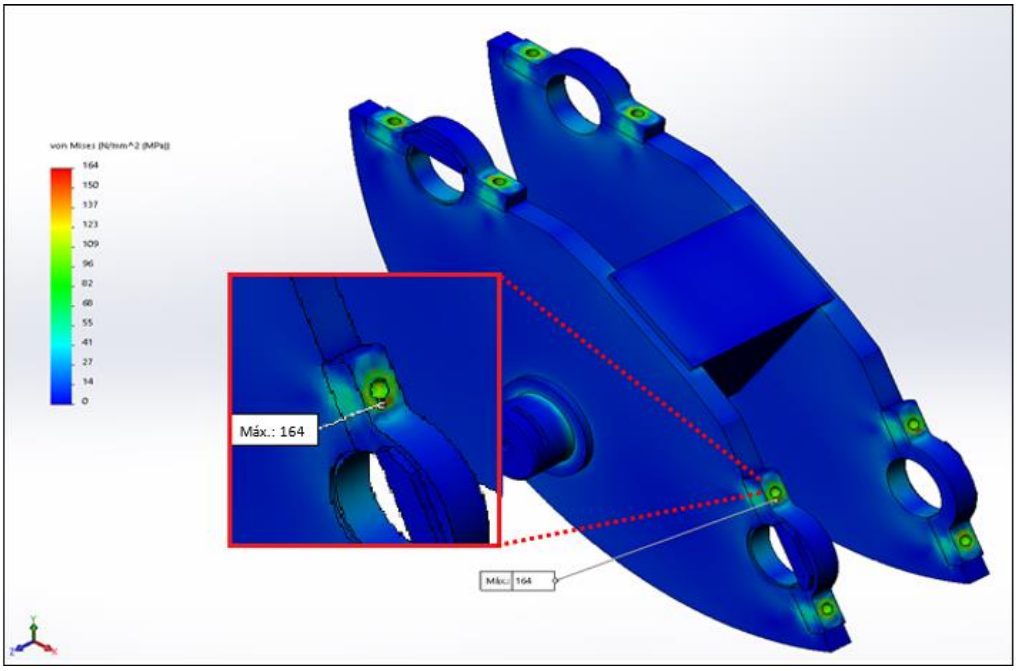

Figure 10. Maximum tension by the Von Mises criterion of 164 MPa acting on the trunnion.

2.2.2 Trunnion and bearing with own weight of the dumper, loaded wagon and free wheels

At this stage the analysis predicts a situation where the equalizing wheels are unlocked during the rotation of the dumper, additionally considering that a tangential force of the rails in the equalizing wheels of 425.4 kgf per wheel is being transmitted.

According to simulation results, it was verified that there is a uniform distribution of tensions in the structure of the piece, of moderate intensity. As expected, the higher stresses are concentrated in the bearing and balance attachment region, with a strength of 199 MPa. Despite the increase in normal stresses in the structure, the utilization rate (u) is approximately 1.7 being within the conditions determined by standard [2]. In the rest of the structure tensions are lower, as shown in figure 11.

2.2.3 Trunnion and bearing with own weight of the dumper, wagon loaded and wheels locked

In this stage a hypothetical situation is proposed, in which the two equalizer wheels of the rocker are locked in the operating rotation, exerting a tangential force of the rails in the equalizing wheels of 1276.13 kgf/wheel, which require the rocker frame to work at maximum possible charging.

In this evaluation, it was verified that there was a slight increase in stress distribution uniformly in the workpiece structure, where the highest stresses remain concentrated in the same bearing and balance setting region with an acceptable intensity of 200 MPa (Figure 12). The utilization rate (u) is approximately 1.7 being within the conditions determined by norm [2].

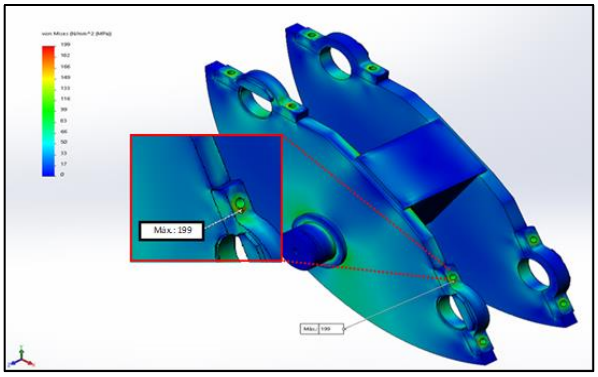

Figure 11. Maximum tension - Von Mises of 199 MPa acting on the trunnion.

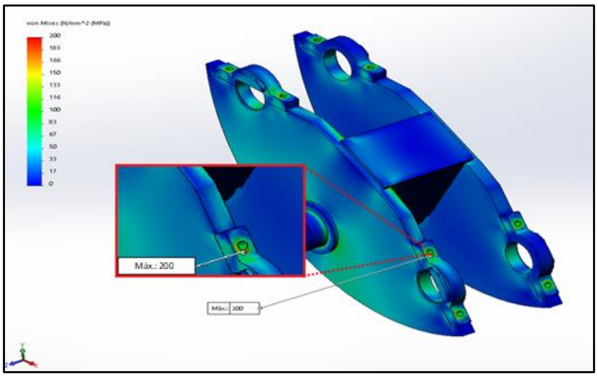

Figure 12. Maximum tension - Von Mises of 200 MPa acting on the trunnion.

2.3 Fatigue Failure Verification

This evaluation aims to identify parts of the structure that are subject to failure due to fatigue, which can generate cracks in the structure. Thus, to identify points that may arise premature failure in the structure of the trunnion even with static tensions smaller than those previously verified.

In the point with higher stress concentrations identified in previous topics, it is necessary to evaluate fatigue criteria due to its requirement of structural integrity and to be the point where there will be changes proposed in the project.

The results obtained from the characterization of the simulation in the best case where only the weight of the dumper and equalizer wheels are considered unlocked in relation to the worst case, where the weight of the dumper with loaded wagon and locked equalizer wheels is considered.

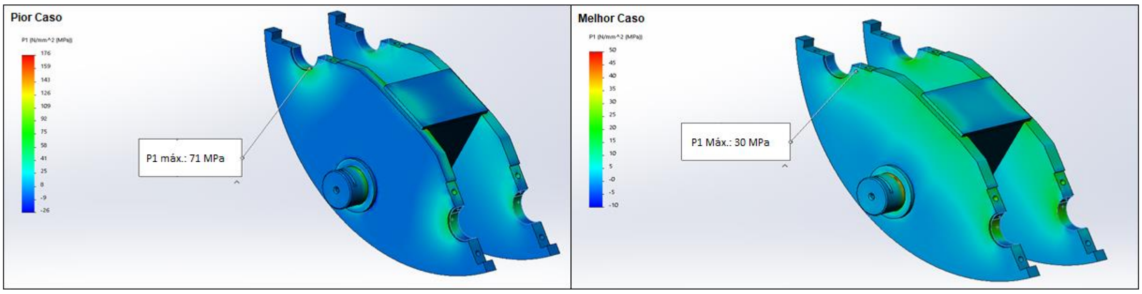

Figure 13. Representation of the 1st principal in traction in the worst and in the best case.

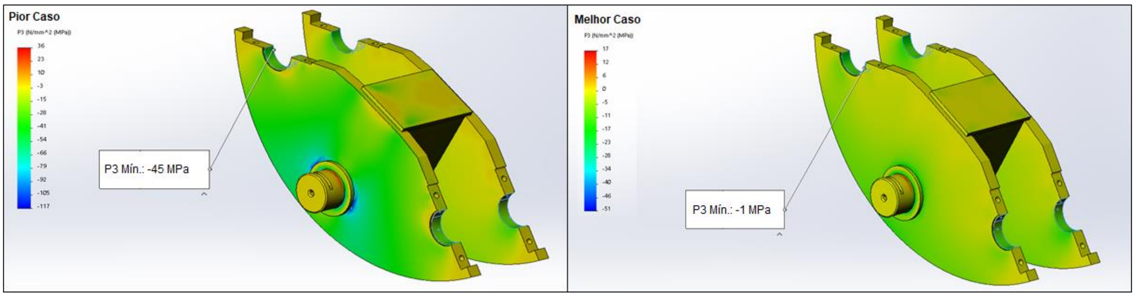

Figure 14. Representation of the 3rd principal in compression in the worst case.

Table 2 list the consolidation of the values of the tensions obtained in the bearing region of the equalizing wheel, which will be used in the calculation of fatigue.

Table 2. Consolidation of the stresses P1 and P3 acting on the trunnions.

Parameters	Tensões Atuantes no Balancim		
	Dumper + Free Wheel	Dumper + Loaded Wagon + Locked Wheel + Axial Force	Difference (%)
P1	30	71	237
P3	-1	-45	-2

2.3.1 Determination of the number of cycles

For a useful life of 25 years on the trunnion, considering that the rotary car dumper has a capacity of 7000t / h and at each cycle the net load will be 160t (equivalent to the mass of a pair of loaded wagons) the number of cycles will be:

$$Cic_{25} = \frac{7000}{160} * 24 * 365 * 25 \rightarrow Cic_{25} = 9581250 \; cycles$$

2.3.2 Determination of the Spectrum Class of the component

To determine the spectrum class, it is necessary to calculate the 'spectrum factor' according to FEM [2], item 2-1.4.3, which characterizes the magnitude of the forces acting on the component during its useful life. In this study it was considered that 50% of the cycles the wagons perform a discharge, and 50% of the cycles it returns to the original position. To determine the '*ksp*' a 100-cycle machine operation was used.

$$k_{sp} = \left(\frac{\sigma_1}{\sigma_{max}}\right)^c * \frac{n_1}{N} + \left(\frac{\sigma_2}{\sigma_{max}}\right)^c * \frac{n_2}{N} \rightarrow k_{sp} = \left(\frac{71}{71}\right)^3 * \frac{50}{100} + \left(\frac{30}{71}\right)^3 * \frac{50}{100}$$
$$\rightarrow k_{sp} = 0{,}5 + 0{,}03772 \rightarrow k_{sp} = 0{,}5377$$

Where 'c' is the factor that characterizes the slope of the Wöhler curve, and that according to the FEM [2], it must be adopted with the absolute value of '3'.

The 'spectrum class' was obtained from the value of 'ksp', by consulting the FEM table in section 2.10 [2], item 2-1.4.3; then the 'spectrum class' for the two cases will be classified as 'P4'.

2.3.2 Determination of Class Component Use

Before determining the class of use of the component, it is necessary to calculate the 'class of use of groups' through the 'time of operation of the component'. For this, it was calculated considering a number of 9581250 cycles of 79s duration, predicting a conservative case in the operation of 24h/day, 365 days/year and 25 year life. Therefore:

$$t_o = 24 * 365 * 25 \Rightarrow t_o = 219000{,}0h$$

Then, according to the table in item 2-1.2.2 [2], the group utilization class is A8, and was verified in section 2-1.4.2 of the FEM [2], comparing directly the number of cycles expected, and then found the B10 class.

Therefore, crossing the spectrum of P4 class with the class of use of the component in Item B10 Table 2-1.4.4 FEM [2], gets the rating group E8.

The constructive characterization of the component, based on the reality of the area, will be considered the non-welded component to determine the notch factor. Thus, the structure of the trunnion was classified in category 'W0' considering as the most severe case of operation, and 'W1' to the case of lower structural requirement, as can be seen in FEM [2], item 3-4.2 and 3 -4.5.2.

From the characterization of the component, its category (W0 and W1) and the Utilization Class (E8), the maximum reference tension (tension W) in the region of the hole will be determined, according to the table in item 3-4.5.1.1 of the FEM [2], indicates that this maximum reference voltage will be:

- $\sigma_W = 102{,}0MPa$; for the own weight of the dumper and free equalizer wheels;

- $\sigma_W = 120{,}0MPa$; for the own weight of the dumper, loaded wagon and free and locked equalizer wheels;

To determine the maximum stress allowed by the fatigue, the correction factor 'k' must be calculated, which correlates the minimum tension with the maximum tension applied to each load case studied previously and is given by the following formula, according to FEM item 3- 4.4 [2].

Dumper + Free Equalizer Wheels

$$\sigma_1 = \sigma_{max} = 30\ MPa;\ \sigma_2 = \sigma_{min} = -1\ MPa;$$

$$k = \frac{\sigma_{min}}{\sigma_{max}} \rightarrow k = \frac{-1}{30} \rightarrow k = -0{,}0333$$

Dumper + Loaded Wagon + Locked Equalizer Wheels + Axial Force

$$\sigma_1 = \sigma_{max} = 71\ MPa;\ \sigma_2 = \sigma_{min} = -45\ MPa;$$

$$k = \frac{\sigma_{min}}{\sigma_{max}} \rightarrow k = \frac{-45}{71} \rightarrow k = -\ 0{,}6338$$

According section 3-4.5.1.1 of the FEM [2], the formulas that should be used in case of 'k≤0' for components subject to traction and compression, in the various loading conditions, are described below. Immediately thereafter, it is carried out by applying specific security coefficient item 3.4.5 EMF [2], which corresponds to 75% of the rated tension, also yielding the following values for the permissible stress, in many cases studied.

For traction calculation:

$$\sigma_t = \sigma_W * \frac{5}{3-(2*k)}\ ;\ \sigma_a = 0{,}75 * \sigma_t \quad (1)$$

For compression calculation:

$$\sigma_c = \sigma_W * \frac{2}{1-k}\ ;\ \sigma_a = 0{,}75 * \sigma_c \quad (2)$$

The calculation results are presented in Table 3.

Table 3. Consolidation of traction and compression tensions according to FEM.

Description	Inputs	Tensions Permissible for Fatigue k≤0 (MPa)	Allowable stress (MPa)
Traction	Dumper Mass+ Free Wheel	166	125
	Dumper Mass + Loaded Wagon + Locked Wheel + Axial Force	141	105
Compression	Dumper Mass+ Free Wheel	197	148
	Dumper Mass + Loaded Wagon + Locked Wheel + Axial Force	147	110

According to Table 3 it is shown that for the worst case:

- **For Traction:**

$$\sigma_a = 105{,}0\ MPa > \sigma_m = 71{,}0\ MPa$$

The structure meets the Technical Specification to useful life of 25 years

- **For Compression:**

$$\sigma_a = -110{,}0\ MPa > \sigma_m = -45{,}0\ MPa$$

$The\ structure\ meets\ the\ Technical\ Specification\ to\ useful\ life\ of\ 25\ years$

3 CONCLUSION

The results of static analysis showed that changing trunnions design will not affect the structural behavior when subjected to loads during operation of the equipment.

According to FEM 2 131/2 132 [2] the results of the fatigue calculations using analysis of static, showed that in most severe operating conditions (which was considered the loaded weight of the loaded wagon and the equalizing wheels locked during the operational turn) the structure meets the technical specification for a useful life of 25 years in operation, and may fail due to fatigue after the end of this time.

Therefore, the structural alteration in the constructive design of the trunnions is considered feasible, since with the implemented improvements the maintenance conditions of the asset will be increased and consequently, the equipment downtime for maintenance work on equalizing wheels is decreased.

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